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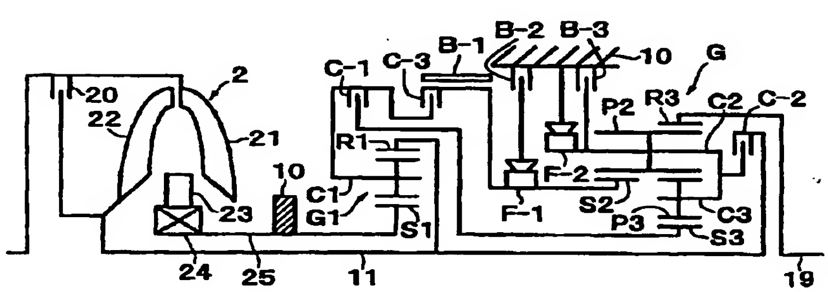
(54) **Automatic transmission for vehicle**

(57) An automatic transmission comprises a reduction planetary gear (G1), two clutches (C-1,C-3) transmitting a decelerated rotation from the reduction planetary gear (G1), and a planetary gear (G) set comprising two elements (S2,S3) to which the decelerated rotation is inputted through the two clutches (C-1,C-3). The reduction planetary gear (G1) and the two clutches (C-1,C-3) are arranged on one side of the planetary gear set (G). The clutch (C-3) is closer to the planetary gear set than the clutch (C-1). The input side member of the clutch (C-1) is connected to an output element (C1) of the reduction planetary gear (G1) and the input side member of the clutch (C-3). The output side member of the clutch (C-1) is extended through the radial inner space of the clutch (C-1) and connected to one of the two elements of the planetary gear set (G). Then, no member is arranged in the radial outer side of the both clutches (C-1,C-3), and the both clutches (C-1,C-3) are radially enlarged.

FIG. 1



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Description

BACKGROUND OF THE INVENTION

1. Field of the Invention

[0001] The invention relates to an automatic transmission for a vehicle. More specifically, it relates to the arrangement of the each transmission structural element in the gear train.

2. Description of the Related Art

[0002] The automatic transmission for a vehicle is required to multiply the gear ratios for an improvement of a driving performance and a fuel consumption. According to the requirement, the transmission mechanism tends to transit from the mechanism having four forward speeds to the mechanism having five forward speeds. For achieving further multiplication of the gear stages in the limited mounting space of a vehicle, a reduction of a number of elements of the gear train and a simplification of the mechanism are needed. A gear train, which achieves six forward speeds and one reverse speed with a planetary gear set having minimum shift elements, is suggested in U.S.P. 5,106,352. In this gear train, six forward speeds are established by that an input rotation inputted to the shift mechanism and two decelerated rotations decelerated from the input rotation are inputted to the planetary gear set having four shift elements.

[0003] In the aforementioned gear train, a number of shift elements for establishing gear stages and a number of needed clutches and brakes are reasonable. However, the gear train has problems which have to be practically improved. In the gear train, two clutches are needed for inputting the decelerated rotations from the reduction planetary gear to two different elements of the planetary gear set. These clutches need larger torque capacities than a clutch inputted a non-decelerated rotation. In the aforementioned structure, the two clutches for transmitting the decelerated rotations are arranged in one side of the reduction planetary gear, and a connecting member for connecting one clutch to one element of the planetary gear set is arranged in radial outer position of the other clutch. Therefore, the diameter of the other clutch is limited, and it is difficult to obtain the capacity by increasing the diameter. As a result, it is needed to increase a number of the structural elements of the frictional member for obtaining the capacity which is usually set based on the diameter of the frictional member and a number of the structural elements. Therefore, the size and the weight of the transmission are increased because of increase of the axial size.

SUMMARY OF THE INVENTION

[0004] In view of the above problems associated with the related art, an object of the invention is to provide an automatic transmission having a shift mechanism, which establishes multiple gear stages by inputting two decelerated rotations to a planetary gear set, and obtaining enough torque capacity of two decelerated rotation input clutches without increasing the size of the shift mechanism.

[0005] For achieving the aforementioned object, in the invention, an automatic transmission for a vehicle comprises a reduction planetary gear, two clutches transmitting a decelerated rotation which is transmitted through at least the reduction planetary gear, and a planetary gear set inputted the decelerated rotation which is transmitted through the two clutches. The automatic transmission establishes multistage gear ratios by these. In the automatic transmission, the reduction planetary gear and first and third clutches, which are the two clutches, are arranged in one side of the planetary gear set. The first and third clutches input the decelerated rotation transmitted through the reduction planetary gear to two different elements of the planetary gear set. The third clutch is arranged in closer position to the planetary gear set than the first clutch. An input side member of the first clutch is connected to an output element of the reduction planetary gear and an input side member of the third clutch, and an output side member of the first clutch is extended through an inner space of the third clutch and connected to one of the two elements of the planetary gear set.

[0006] Then, it is effective that one element of the reduction planetary gear is fixed to a boss portion extended from the transmission case.

[0007] Further, it is effective that the transmission case comprises a support wall, the first clutch is arranged on the boss portion, and the third clutch is arranged in a neighbor position of the support wall.

[0008] Especially, in case the transmission comprises a plural of parallel shafts, it is effective that the output member is a counter gear, and the counter gear is arranged in the other side of the support wall against the third clutch.

[0009] In the aforementioned structure, it is effective that the support wall comprises a second cylindrical portion supporting the counter gear.

[0010] Further, it is effective that the third clutch is supported on a first cylindrical portion axially extended from the support wall.

[0011] Further, it is effective that the input side member of the first clutch is a clutch drum, of which a hydraulic servo is formed in the inside, and arranged in order that a cylinder of the hydraulic servo opens to the reduction planetary gear.

[0012] Further, it is effective that the output side member of the third clutch is a clutch drum, of which a hydraulic servo is formed in the inside, arranged in order

that a cylinder of the hydraulic servo opens to the opposite side against the support wall, and connected to one element of the planetary gear set.

[0013] Further, it is effective that the first engaging member is structured from a band brake in which the outer surface of the clutch drum of the third clutch is an engaging surface engaged by a band.

[0014] Further, it is effective that the transmission case comprises a support wall, the first clutch is arranged in one side of the support wall, and the third clutch is arranged in the other side.

[0015] Further, it is effective that the first and third clutches are supported on a cylindrical portion axially extended from the support wall.

[0016] In case the transmission establishes six forward speeds, it is effective that a first element of the planetary gear set is connected to the output side member of the first clutch, a second element is connected to an output side member of the third clutch and is possible to engage with a transmission case with a first engaging member, a third element is connected an output side member of a second clutch, which is inputted a non-decelerated rotation, and is possible to engage with a transmission case with a second engaging member, and shift rotations are outputted from a fourth element.

[0017] Further, it is effective that the second clutch is arranged in neighbor position of the reduction planetary gear.

[0018] Further, it is effective that the second clutch is arranged in the inner side of the frictional members of the first and third clutches.

[0019] Further, it is effective that an input side member of the second clutch is a clutch drum, and the clutch drum is an input member inputting a rotation to the reduction planetary gear.

[0020] Further, it is possible to structure in order that the output side member of the second clutch is a clutch drum, and the clutch drum is connected through an intermediate shaft to the third element of the planetary gear set.

[0021] Further, it is effective that a frictional member, which connects between the input side member and the output side member, of the second clutch is arranged in the radial outer position of the reduction planetary gear.

[OPERATION AND EFFECTS OF THE INVENTION]

[0022] In the structure of the invention, the input side member of the third clutch is connected through the input side member of the first clutch to the output element of the reduction planetary gear, and the output side member of the first clutch is extended through the inner space of the third clutch and connected to one of the two elements of the planetary gear set. Therefore, a member, which transmits the output rotation from the reduction planetary gear through the first and third clutches to the planetary gear set, is not needed to be arranged in the radial outer position of the both clutches. As a result, the diameter of the first and third clutches, which transmit the torque amplified through the reduction planetary gear, is possible to be increased. Therefore, it becomes easy to obtain the enough torque transmission capacity without increasing axial size of the frictional member caused by increasing a number of the structural elements of the frictional member. As a result, the transmission is structured in compact.

[0023] One element of the reduction planetary gear is fixed to the transmission case so that a support, which is particularly arranged for fixing, is not needed.

[0024] A hydraulic pressure is applied to the hydraulic servo of the third clutch from the support wall closed to the hydraulic servo. Therefore, a hydraulic path for applying the hydraulic pressure to the hydraulic servo is shortened so that a responsiveness to the application of the hydraulic pressure is improved. Further, the axial size of the transmission is increased by arranging the support wall, however, the axial size of the first and third clutches are able to be reduced because the enough torque transmission capacity is obtained by increasing the diameters of the both clutches. The diameters are able to be increased because no member is extended in the radial outer space of the first and third clutches. Therefore, the increase of the axial size of the transmission is minimized.

[0025] The support wall for supporting the counter gear and the support wall for applying the hydraulic pressure to the third clutch are common support wall so that the axial size of the transmission is reduced.

[0026] The counter gear, which transmits the torque amplified by a shift, is certainly supported by the support wall.

[0027] A hydraulic pressure is directly applied to the hydraulic servo of the third clutch from the cylindrical portion of the support wall without through an another rotating member. Therefore, a number of seal rings, which are arranged for sealing the hydraulic path, is reduced so that a sliding load caused by the seal rings is reduced. As a result, a transmission efficiency of the transmission is improved.

[0028] The input side member of the first clutch is the clutch drum, and the drum is arranged in mostly radial outer position of the shift mechanism. Therefore, it is easy to detect an input rotation needed for a transmission control without arranging a sensor in a deep position of the transmission case. Further, the hydraulic servo is faced to the reduction planetary gear side so that a complicated arrangement of the connecting member, which is caused in case the frictional member is arranged in the radial outer space of the reduction planetary gear, is prevented.

[0029] The hydraulic servo of the third clutch is faced to opposite side of the support wall so that a complicated arrangement of the connecting member, which is caused in case the frictional member is arranged in

the radial outer space of the reduction planetary gear, is prevented. Further, the clutch drum is supported by the support wall and connected to the input element of the planetary gear set without a complicated arrangement of the connecting member.

[0030] The first engaging member is the band brake. The band brake needs the arrangement space which is pretty small in the radial direction. Therefore, the radial outer space of the third clutch is obtained and the diameter of the clutch is able to be increased even by radially superposing the first engaging member and the third clutch which are connected to the second element of the planetary gear set. The torque transmission capacity is obtained by increasing the diameter of the clutch. As a result, the axial size of the clutch is reduced.

[0031] Hydraulic pressures are applied to the hydraulic servos of the first and third clutches from the support wall closed to the both clutches. Therefore, hydraulic paths for applying the hydraulic pressures to the hydraulic servos of the both clutches are shortened, and the lengths of the hydraulic paths are almost equable. Therefore, a responsiveness of the each clutch to the application of the hydraulic pressure is improved, and control characteristics of the clutches are accommodated. Further, the axial size of the transmission is increased by arranging the support wall, however, the axial size of the first and third clutches are able to be reduced because the enough torque transmission capacity is obtained by increasing the diameters of the both clutches. The diameters is able to be increased because no connecting member is extended in the radial outer space of the first and third clutches. Therefore, the increase of the axial size of the transmission is minimized.

[0032] The first and third clutches are supported by the support wall integrated with the transmission case. Therefore, the supports are more stabled comparing with a case in which the clutches are supported through a movable member such as a rotating shaft to the transmission case. Further, hydraulic pressures are directly applied to the hydraulic servos of the first and third clutches from the support wall without through an another rotating member. Therefore, a number of seal rings, which are arranged for sealing the hydraulic paths, is reduced so that a sliding load caused by the seal rings is reduced. As a result, a transmission efficiency of the transmission is improved.

[0033] A shift mechanism establishing proper six gear ratios is achieved, and the mechanism is compact and able to achieve the aforementioned effects.

[0034] The second clutch and the first and third clutches, which have different torque transmission capacity, are arranged together in the neighbor position of the reduction planetary gear. The frictional member, of which diameter is able to be reduced because of the small torque transmission capacity, structuring the second clutch is arranged in the radial inner position of the frictional member of the another clutch. Therefore, the arrangement space is reduced and the whole shift mechanism is compact.

[0035] The input side member of the second clutch is common with the input member connected to the reduction planetary gear so that a number of power transmission members which radially across in the shift mechanism is reduced. Therefore, the axial size of the shift mechanism is reduced.

[0036] The clutch drum, which is the output side member, of the second clutch is connected to the intermediate shaft, and the clutch hub, which is the input side member, is positioned in the reduction planetary gear side. Therefore, the structure for inputting the non-decelerated rotation to the clutch hub is simplified.

[0037] The input side member of the second clutch is the clutch hub, and the hub is common with the input element of the reduction planetary gear. Therefore, the arrangement space for the second clutch is reduced. As a result, the shift mechanism is compact.

Brief Description of the Drawings

[0038] The invention will be described in conjunction with the following drawings in which like features are designated with like reference characters, and wherein:

Fig. 1 is a schematic diagram illustrating a gear train of the first embodiment of an automatic transmission applied the invention;

Fig. 2 is a velocity diagram for the gear train;

FIG. 3 is a table showing the operation of the gear train, the established gear ratios and the gear ratio steps;

FIG. 4 is an axial cross sectional diagram simply illustrating a shift mechanism portion of the gear train;

FIG. 5 is an axial cross sectional diagram simply illustrating the second embodiment in which the shift mechanism portion is changed;

FIG. 6 is an axial cross sectional diagram simply illustrating the third embodiment in which the shift mechanism portion is further changed;

FIG. 7 is an axial cross sectional diagram simply illustrating the fourth embodiment in which the shift mechanism portion is further changed;

FIG. 8 is an axial cross sectional diagram simply illustrating the fifth embodiment in which the shift mechanism portion is further changed;

FIG. 9 is an axial cross sectional diagram simply illustrating the seventh embodiment in which the shift mechanism portion is further changed;

FIG. 10 is a schematic diagram illustrating the seventh embodiment in which the shift mechanism portion is further changed;

FIG. 11 is an axial cross sectional diagram simply illustrating the shift mechanism of the seventh

embodiment;

Fig. 12 is an axial cross sectional diagram simply illustrating the eighth embodiment in which the shift mechanism portion is further changed;

Fig. 13 is an axial cross sectional diagram simply illustrating the ninth embodiment in which the shift mechanism portion is further changed;

Fig. 14 is an axial cross sectional diagram simply illustrating the tenth embodiment in which the shift mechanism portion is further changed;

Fig. 15 is an axial cross sectional diagram simply illustrating the eleventh embodiment in which the shift mechanism portion is further changed;

Fig. 16 is an axial cross sectional diagram simply illustrating the twelfth embodiment in which the shift mechanism portion is further changed;

Fig. 17 is an axial cross sectional diagram simply illustrating the thirteenth embodiment in which the shift mechanism portion is further changed;

FIG. 18 is a schematic diagram illustrating the gear train of the fourteenth embodiment of the automatic transmission applied the invention;

Fig. 19 is an axial cross sectional diagram simply illustrating the shift mechanism of the fourteenth embodiment; and

Fig. 20 is an axial cross sectional diagram simply illustrating the fifteenth embodiment in which the shift mechanism portion is further changed.

Detailed Description of the Invention

[0039] The invention will become more apparent from detailed description of preferred embodiments with reference to the accompanying drawings. Fig.1 illustrates a gear train of the first embodiment of the automatic transmission for a vehicle applied the invention with a schematic diagram. This transmission is a parallel type transmission for a front engine/rear drive (FR) vehicle, and establishes multiple gear stages with a reduction planetary gear G1, three clutches C-1, C-3 including two clutches which transmit a decelerated rotation transmitted through the reduction planetary gear G1, and a planetary gear set G inputted the decelerated rotation transmitted through the two clutches which are a first clutch C-1 and a third clutch C-3.

[0040] In the planetary gear set, a first element S3 is connected to the output side member of the first clutch C-1, a second element S2 is connected to the output side member of the third clutch C-3 and is able to be engaged with a transmission case 10 with a first engaging member (B-1, F-1, B-2), a third element C2 (C3) is connected to the output side member of the second clutch C-2 inputted a non-decelerated rotation and is able to be engaged with the transmission case 10 with a second engaging member (B-3, F-2), and a fourth element R3 (R2) is connected to a output shaft 19 as a output member. In this automatic transmission having the structure of the gear train, the each clutch and engaging member is selectively engaged/disengaged by applying/draining a hydraulic pressure to/from the each hydraulic servo of the clutches and engaging member by a hydraulic control system which is not shown in the figure so that gear ratios having six forward speeds and one reverse speed are established.

[0041] The gear train of this embodiment will be described in further detail as follows. With reference to Fig. 1, in this automatic transmission, a torque converter 2 having a lock-up clutch 20 and connected to an engine, not shown in the figure, is arranged in the mostly front position, and the shift mechanism is arranged in the rear position. The torque converter 2 comprises a pump impeller 21, turbine runner 22, a stator 23 arranged between them, a one-way clutch 24 engaging the stator 23 with the transmission case in the one-way direction, and a stator shaft 25 fixing the inner race of the one-way clutch to the transmission case 10.

[0042] The planetary gear set G structuring the main part of the shift mechanism is a ravegneaux type gear set comprising a small sun gear S3 as the first element having small diameter, a large sun gear S2 as the second element having large diameter, carriers C1, C3 integrated each other as the third element supporting a pair of pinions P2, P3. The pinions P2, P3 are engaged with each other, the pinion P2 is a long pinion engaged with the large sun gear S2 and a ring gear R3 (R2), and the pinion P3 is a short pinion engaged with the small sun gear S3. It should be noted that the ring gear R3 (R2) and the carriers C2, C3 are theoretically structured from two different elements individually, however, the ring gear is actually one element, and the carrier is one element because the carriers are integrated each other. Therefore, in the description, regarding to the former, according to the positional relationship with the sun gears S2, S3, a notation R2 is put in case the ring gear is arranged in the radial outer position of the sun gear S2, and a notation R3 is put in case the ring gear is arranged in the radial outer position of the sun gear S3. Further, regarding to the later, one of notations is omitted.

[0043] The reduction planetary gear G1 is structured from a simple planetary gear. A ring gear R1 as the input element is connected to an input shaft connected to the turbine runner of the torque converter, a carrier C1 as the output element is connected to the input side member of the first clutch C-1 and connected through the input side member to the input side member of the third clutch C-3, and a sun gear S1 as one element acting as a reaction element is fixed to the transmission case 10.

[0044] The automatic transmission thus structured shifts according to a vehicle load within gear stages corresponding to a shift range selected by a driver by an electronic control system and a hydraulic control system which are not shown in the figure. Fig. 3 shows the gear stages, which are established by engagement and disengagement of each clutch and brake, with a table. In

this table, a symbol ○ member engagement, a blank member disengagement, a symbol △ member engagement which is performed only in case an engine brake is established, and a symbol ● member engagement which is not directly operated for establishing the gear stages. Fig. 2 shows the gear stages, which are established by engagement and disengagement of each clutch and brake, and a relationship with the rotational ratio of each shift element in each gear stage with a velocity diagram. In this diagram, a symbol ● member engagement.

[0045] As shown in Fig. 2 and Fig. 3, a first speed (1ST) is established by engagement of the clutch C-1 and the brake B-3. It should be noted that, in this embodiment, as shown in the operational table, the automatic engagement of the one-way clutch F-2 is used instead of engagement of the brake B-3. A reason for using this engagement and a reason that this engagement is equivalent to engagement of the brake B-3 will be described later. In this gear stage, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 shown in Fig. 1 through the reduction planetary gear G1, is inputted to the small sun gear S3 through the clutch C-1, the carrier C3, which is engaged with the transmission case 10 by engagement of the one-way clutch F-2, acts as a reaction element, and the decelerated rotation of the ring gear R3 having the largest reduction ratio is outputted to the output shaft 19.

[0046] Next, a second speed (2ND) is established by engagement of the clutch C-1, and engagements of the one-way clutch F-1 and the brake B-2, which is engaged for validating the engagement of the one-way clutch F-1, which are equivalent to engagement of the brake B-1. It should be noted that the reason of that the engagements of the one-way clutch F-1 and the brake B-2 are equivalent to the engagement of the brake B-1 will be mentioned later. In this gear stage, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 through the reduction planetary gear G1, is inputted to the small sun gear S3 through the clutch C-1, the large sun gear S2, which is engaged with the transmission case 10 by engagements of the brake B-2 and the one-way clutch F-1, acts as a reaction element, and the decelerated rotation of the ring gear R3 is outputted to the output shaft 19. The reduction ratio in this case is less than the reduction ratio in the first speed (1ST) as shown in Fig. 2.

[0047] Further, a third speed (3RD) is established by concurrently engagements of the clutch C-1 and the clutch C-3. In this case, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 through the reduction planetary gear G1, is concurrently inputted to the large sun gear S2 and the small sun gear S3 through the clutch C-1 and the clutch C-3 individually so that the planetary gear set G is in the directly connected state, the rotation of the ring gear R3, which is same with the input rotation inputted to the both sun gears, is outputted to the output shaft 19 as the rotation decelerated from the rotation of the input shaft 11.

[0048] Further, a fourth speed (4TH) is established by concurrently engagements of the clutch C-1 and the clutch C-2. In this case, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 through the reduction planetary gear G1, is inputted to the small sun gear S3 through the clutch C-1 on one side, the rotation, which is not reduced and inputted from the input shaft 11 through the clutch C-2, is inputted to the carrier C3 on the other side, and the middle rotation between the two input rotations, which is the rotation of the ring gear R3 barely decelerated from the rotation of the input shaft 11, is outputted to the output shaft 19.

[0049] Next, a fifth speed (5TH) is established by concurrently engagements of the clutch C-2 and the clutch C-3. In this case, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 through the reduction planetary gear G1, is inputted to the small sun gear S2 through the clutch C-3 on one side, the non-decelerated rotation, which is not reduced and inputted from the input shaft 11 through the clutch C-2, is inputted to the carrier C2 on the other side, and the rotation, which is barely accelerated from the rotation of the input shaft 11, of the ring gear R3 is outputted to the output shaft 19.

[0050] Then, a sixth speed (6TH) is established by engagement of the clutch C-2 and the brake B-1. In this case, the non-decelerated rotation from the input shaft 11, which is not reduced, is inputted to only the carrier C2 through the clutch C-2, the sun gear S2, which is engaged with the transmission case 10 by engagement of the brake B-1, acts as a reaction element, and the rotation of the ring gear R3, which is further accelerated, is outputted to the output shaft 19.

[0051] A reverse speed (REV) is established by engagements of the clutch C-3 and the brake B-3. In this case, the decelerated rotation, which is decelerated from the rotation of the input shaft 11 through the reduction planetary gear G1, is inputted to the sun gear S2 through the clutch C-3, the carrier C2, which is engaged with the transmission case 10 by engagement of the brake B-3, acts as a reaction element, and the rotation of the ring gear R3, which is a reverse rotation, is outputted to the output shaft 19.

[0052] As shown with the clearance in the up/down direction between symbols ○ which show the speed ratios of the ring gear R3, R2 on the velocity diagram in Fig. 2, the each gear stage thus established has a proper speed step having a relatively equivalent interval to the each gear stage. This relationship is shown with gear ratios and steps between gear ratios in Fig. 3 by specifically setting values and quantitatively showing. The gear ratio in this case is a value in case a teeth number ratio $\lambda 1$ between the sun gear S1 and the ring gear R1 of the reduction planetary gear G1 equals to 0.556, a teeth number ratio $\lambda 2$ between the large sun gear S2 and the ring gear R3 of the planetary gear set

G equals to 0.458, and a teeth number ratio $\lambda 3$ between the small sun gear S3 and the ring gear R3 equals to 0.375, the width of the gear ratios is equals to 6.049.

[0053] Here, the relationship between the one-way clutch F-2 and the brake B-3 and the relationship between the one-way clutch F-1 and the both brakes (B-1, B-2) mentioned above will be described. As shown in the relationship of engagement/disengagement of both brakes (B-1, B-2) in the first speed and the second speed, one of these brakes is disengaged and the other brake is engaged concurrently in the up/down shift between these gear stages. That is, these brakes are frictional engagement elements which are changed states of them each other. In this changing states of the frictional engagement elements, a precise concurrent control for the engagement pressure and the disengagement pressure of the hydraulic servos which control the brakes is needed. For performing the control, an addition of control valves and a complication of a hydraulic circuit are needed. In this embodiment, the engagement direction of the one-way clutch F-2 is set to be same with the supporting direction for the reaction torque in the first speed, then the one-way clutch F-2 is substantially operated same function with the engagement of the brake B-3 and an automatic disengaging function, which is not operated by the brake, by using that the reaction torque received by the carrier C2 is reversed between the first speed and the second speed. That is, in the first speed, the carrier C2 is engaged by the one-way clutch F-2 instead of the brake B-3. It should be noted that, in an engine coasting state in which the wheels drive the engine, the engagement of the brake B-3 is needed, as shown with the symbol Δ in Fig. 3, for achieving an engine brake because the direction of the reaction force received by the carrier C2 is reversed from the case of an engine drive state. Therefore, it is possible to be structured so that the first speed is established by engagement of the brake B-3 without arranging the one-way clutch for establishing the gear stage.

[0054] The same relationship with the aforementioned relationship is established in a case of the sun gear S2. In this case, the engagement direction of the one-way clutch F-1 is set as the same direction in which the sun gear S2 supports the reaction torque in the second speed so that the one-way clutch F-1 performs substantially equivalent function with the engagement of the brake B-1. The function of the sun gear S2 is different from the function of the carrier C2 (C3). That is, the sun gear S2 is engaged with the case 10 for achieving the engine brake in the second speed and for establishing the sixth speed. Therefore, the brake B-1 is needed. Further, as shown in the velocity diagram in Fig. 2, the sun gear S2 rotates in reverse direction from the direction of the input rotation in the first speed, but rotates in same direction with the direction of the input rotation in the gear stages higher than the third speed. Therefore, the one-way clutch F-1 is not able to be directly connected to a stationary member. Therefore, the one-way clutch F-1 and the brake B-2 are arranged in series so that the engagement state of the one-way clutch F-1 is effectively operated.

[0055] Next, Fig. 4 illustrates the shift mechanism portion of the aforementioned gear train in further detail with a cross sectional view. The each structural element, which is described before with reference to the schematic diagram, is putted the same reference numeral or notation and the description is omitted. Detail portions, which are not able to be shown in the schematic diagram, will be described as follows. It should be noted that, through the specification, a term clutch is a generic name of a frictional member, a drum and a hub which are a support member for and a power transmission member, and a hydraulic servo for operating the frictional member to engage and disengage. Further, as regarding brake, in case the brake is multidisc type as same as a clutch, a term brake is a generic name of a frictional member, a hub as a support member and a power transmission member, and a hydraulic servo for operating the frictional member to engage. In case the brake is structured from a band brake, the term brake is a generic name of a brake band, a drum structuring an engagement surface, a hydraulic servo for engaging the brake band.

[0056] The transmission case 10 has a cylindrical shape. The transmission case 10 has a cylindrical boss portion 10b extended from a front end wall portion 10a to the inner side of the transmission case 10 and a cylindrical boss portion 10d extended from a rear end wall portion 10c to the inner side of the transmission case 10. A support wall 10A is fixed to the axial middle portion of the transmission case 10. The support wall 10A comprises a radial wall portion 10e connected to the transmission case 10 and a front cylindrical portion 10f' axially extended from the inner portion of the radial wall portion 10e to the front side. It should be noted that a notation Sn shows an input rotation sensor detecting an input rotation of the shift mechanism for a shift control.

[0057] Next, the input shaft 11, which inputs a power transmitted through the torque converter to the shift mechanism, is divided to a front portion 11A and a rear portion 11B for an accommodation for processing, and the two shafts are integrally connected with each other by closely engaging with spline engagement. The front portion 11A of the input shaft 11 is connected to the turbine runner 22 of the torque converter 2, in the transmission case 10, the front end portion is supported through a bearing on the inner surface of the front end wall portion 10a, and the rear end portion is supported through a bearing on the inner surface of the head portion of the cylindrical boss portion 10b. That is, the front portion 11A is rotatably supported by the transmission case 10. The rear end portion of the rear portion 11B is inserted and supported through a bearing in a trough portion of the output shaft 19, and rotatably supported through the output shaft 19 on the inner surface of the

cylindrical boss portion 10d of the transmission case 10. An input member inputting a rotation to the reduction planetary gear G1 is a flange formed on the portion closed to the rear end portion of the front portion 11A, and the flange is connected to the ring gear R1. Further, a flange is formed on a sleeve member 11C which is stationary fixed on the portion closed to the rear end portion of the rear portion 11B and structures the inner portion of the hydraulic servo 50 of the second clutch C-2. The flange is connected to the second clutch by fixing a clutch drum 51 to the flange.

[0058] Next, the output shaft 19 is rotatably supported through a bearing on the inner surface of the cylindrical boss portion 10d of the transmission case at the front end portion and through a bearing by an extension housing fixed to the hindmost portion of the transmission case 10 at the rear end portion. A connecting portion to the ring gear R3 of the planetary gear set G is a flange formed on the head of the output shaft 19, and a member, which is shaped a drum and connected to the flange, is connected to the ring gear R3.

[0059] The planetary gear set G is arranged on the axial middle portion of the rear portion 11B of the input shaft, the small sun gear S3 is rotatably supported on the rear half portion 11B, further, the large sun gear S2 is rotatably supported on the sun gear S3. The carriers C2, C3 supporting the long pinion P2 and the short pinion P3 are integrated. The front end portion of the carriers is rotatably supported on a shaft portion 11E extended from the sun gear S2 to the front side, and the rear end portion is rotatably supported on the rear portion 11B. In this planetary gear set G, a diameter of a first planetary gear G2 side of the planetary gear set G is different from a diameter of a third planetary gear G3 side having the ring gear R3 because no ring gear is arranged in the radial outer position of the first planetary gear G2 side.

[0060] The reduction planetary gear G1 is arranged on the outer surface of the end portion of the cylindrical boss portion 10b of the transmission case. The sun gear as the stationary element is fixed on the cylindrical boss portion 10b with a spline engagement. The carrier C1 structuring the output element of the reduction planetary gear G1 is supported through a bearing on the cylindrical boss portion 10b at one side.

[0061] According to the basic characteristic of the invention, the reduction planetary gear G1 and the first and third clutches C-1, C-3, which input the decelerated rotation transmitted through the reduction planetary gear G1 to the sun gear S3 and the sun gear S3 of the planetary gear set G, are arranged in one side, that is, in the front side in the embodiment, of the planetary gear set G, and the third clutch C-3 is arranged in the closer position to the planetary gear set G than the first clutch C-1. A clutch drum 31 as an input side member of the first clutch C-1 is connected to the carrier C1 of the reduction planetary gear G1 and an input side member 46 of the third clutch C-3. A clutch hub 36 as an output side member of the first clutch C-1 is extended through the inner space of the third clutch and connected to the sun gear S3 of the planetary gear set G.

[0062] The first clutch C-1 is structured from a frictional member 35 comprising multi-disc type frictional elements and separator plates, the clutch drum 31 as the input side member transmitting a torque to the frictional member 35, the clutch hub 36 as the output side member outputting the torque transmitted by the engagement of the frictional member 35, and a hydraulic servo 30 for engaging the frictional member 35. The clutch drum 31 have cylindrical portions at inner and outer portion. A cylinder of the hydraulic servo 30 is structured between the inner side cylindrical portion and the outer side cylindrical portion having small diameter so that the hydraulic servo 30 is connoted. A large diameter cylindrical portion formed by expanding the outer side cylindrical portion is a support portion for the frictional member 35. The outer side of the separator plates are supported on the inner surface of the large diameter cylindrical portion of the clutch drum 31 with a spline engagement, and the inner side of the frictional elements are supported on the outer surface of the clutch hub 36 with a spline engagement. That is, the frictional member 35 is arranged between the clutch drum 31 and the clutch hub 36. The hydraulic servo 30 is structured from the inner side of the clutch drum 31 as the cylinder, a piston 32 axially slidably inserted in the cylinder, a cancel plate fixed on the inner portion of the clutch drum 31, and a return spring arranged between the piston 32 and the cancel plate.

[0063] In the first clutch C-1 thus structured, the hydraulic servo 30 is arranged in the front side of the reduction planetary gear G1 and in the radial outer side of the cylindrical boss portion 10b, and the frictional member 35 is arranged in the radial outer side of the reduction planetary gear G1. The clutch drum 31 of the first clutch C-1 is arranged in order that the cylinder of the hydraulic servo 30 connoted in the drum is opened to the reduction planetary gear G1 side. The inner side cylindrical portion of the clutch drum 31 is connected to the carrier C1 of the reduction planetary gear G1. The clutch hub 36 is connected to a power transmission member 11D supported through a bearing on the front portion 11A of the input shaft, and connected through the power transmission member 11D to the sun gear S3 of the planetary gear set G.

[0064] The third clutch C-3 is structured from a frictional member 45 comprising multi-disc type frictional elements and separator plates, the clutch hub 46 as the input side member transmitting a torque to the frictional member 45, the clutch drum 41 as the output side member outputting the torque transmitted by the engagement of the frictional member 45, and a hydraulic servo 40 for engaging the frictional member 45. The clutch drum 41 have cylindrical portions at inner and outer portion. A cylinder of the hydraulic servo 40 is structured between the inner side cylindrical portion and the outer

side cylindrical portion having small diameter so that the hydraulic servo 40 is connoted. A large diameter cylindrical portion formed by expanding the outer side cylindrical portion is a support portion for the frictional member 45. The outer side of the separator plates are supported on the inner surface of the large diameter cylindrical portion of the clutch drum 41 with a spline engagement, and the inner side of the frictional elements are supported on the outer surface of the clutch hub 46 with a spline engagement. That is, the frictional member 45 is arranged between the clutch drum 41 and the clutch hub 46. The hydraulic servo 40 is structured from the inner side of the clutch drum 41 as the cylinder, a piston 42 axially slidably inserted in the cylinder, a cancel plate fixed on the inner portion of the clutch drum 41, and a return spring arranged between the piston 42 and the cancel plate.

[0065] In the third clutch C-3 thus structured, the hydraulic servo 40 of the third clutch C-3 is supported through a bearing on the outer surface of the front cylindrical portion 10f' of the support wall 10A in the rear side of the reduction planetary gear G1. The frictional member 45, which is arranged in the radial outer position of the reduction planetary gear G1, is aligned with the frictional member 35 of the first clutch C-1 and arranged in the rear side of the frictional member 35. The clutch drum 41 of the third clutch C-3 is arranged so that the cylinder of the hydraulic servo 40 connoted in the drum is opened to the reduction planetary gear G1 side. The inner cylindrical portion of the clutch drum 41 is connected through the power transmission member 11E, which is inserted in radial outer position of the power transmission member 11D, to the large sun gear S2. The clutch hub 46 is connected to the clutch drum 31 of the first clutch C-1.

[0066] The second clutch C-2, as same as the another clutches, is structured from a frictional member 55 comprising multi-disc type frictional elements and separator plates, the clutch drum 51 as the input side member transmitting a torque to the frictional member 55, the clutch hub 56 as the output side member outputting the torque transmitted by the engagement of the frictional member 55, and a hydraulic servo 50 for engaging the frictional member 55. The clutch drum 51 have cylindrical portion at the outer portion. A cylinder of the hydraulic servo 50 is structured between the sleeve member 11C and the outer side cylindrical portion having small diameter so that the hydraulic servo 50 is connoted. A large diameter cylindrical portion formed by expanding the outer side cylindrical portion is a support portion for the frictional member 55. The outer side of the separator plates are supported on the inner surface of the large diameter cylindrical portion of the clutch drum 51 with a spline engagement, and the inner side of the frictional elements are supported on the outer surface of the clutch hub 56 with a spline engagement. That is, the frictional member 55 is arranged between the clutch drum 51 and the clutch hub 56. The hydraulic servo 50 is structured from the inner side of the clutch drum 51 as the cylinder, a piston 52 axially slidably inserted in the cylinder, a cancel plate fixed on the inner portion of the clutch drum 51, and a return spring arranged between the piston 52 and the cancel plate.

[0067] The hydraulic servo 50 of the second clutch C-2 is supported on the input shaft 11 in the rear side of the planetary gear set G. The frictional member 55 is arranged in the front side of the hydraulic servo 50. The clutch hub 56 is connected to the carrier C2 of the planetary gear set G.

[0068] The brake B-1 structuring one of the first engaging member is a band brake comprising the clutch drum 41 as a brake drum and a band 6 which is selectively engaged with the outer surface of the clutch drum 41 of the third clutch C-3. Generally, in case a brake is a band brake, a moment, which inclines the drum axis, is occurred by the radial load in the engagement of the brake. In this embodiment, however, the brake drum is the clutch drum 41, which is supported on the front cylindrical portion 10f' of the support wall 10A, of the third clutch, that is, the drum is supported by the transmission case in the radial inner position of the band engagement portion so that loads occurred in the brake engagement does not affected to another members. It should be noted that the hydraulic servo of the brake B-1 is omitted to illustrate in the figures.

[0069] The inner race of the one-way clutch F-1 structuring the other one of the first engaging member is connected to the drum 41 of the third clutch C-3. The outer race is integrated with the hub 86 of the brake B-2. The one-way clutch F-1 is arranged in the front side of the first clutch C-1, that is, in the mostly front portion of the shift mechanism. The brake B-2, which engages the outer race with the transmission case 10, is a multi-disc type brake comprising a frictional member 85. The frictional member 85 is structured from frictional elements engaged with the hub 86 and separator plates engaged with an inner spline of the transmission case 10. The hydraulic servo 80 of the brake B-2 comprises the front end wall portion 10a of the transmission case 10 as a cylinder, a piston 82 axially slidably inserted in the cylinder, a retainer fixed to the front end wall portion 10a of the transmission case 10, and a return spring arranged between the retainer and the piston 82.

[0070] The brake B-3 structuring the second engaging member is a multi-disc type brake comprising a frictional member 75. The frictional member 75 comprises multiple frictional elements and separator plates. The separator plates are engaged with an inner spline of the transmission case 10. The frictional elements are engaged with the hub 76 fixed to the carrier C2. The frictional member 75 is arranged in the radial outer space, in which the ring gear is not arranged, of the planetary gear G2 side of the planetary gear set G. The hydraulic servo 70 of the brake B-3 comprises the rear end wall portion 10a and the cylindrical boss portion 10d of the transmission case 10 as a cylinder, a piston 72 axially

slidably inserted in the cylinder, a retainer fixed on the cylindrical boss portion 10d of the transmission case 10, and a return spring arranged between the retainer and the piston 72. The extension portion of the piston 72 extended along with the surrounding wall of the transmission case 10 to the rear end of the frictional member 75 is engaged with a spline of the surrounding wall of the transmission case 10.

[0071] The one-way clutch F-2 structuring the other one of the second engaging member is arranged in parallel with the brake B-3. The inner race is connected to the front end portion of the carrier C2. The outer race is engaged with the transmission case. The one-way clutch F-1 is arranged between the support wall 10A and the planetary gear set G.

[0072] Thus, in the first embodiment, a compact shift mechanism having proper six gear stages is obtained by inputting the decelerated rotation transmitted through the two clutches C-1, C-3 from the reduction planetary gear G1 to the two elements S3, S2 of the ravegneaux type planetary gear set G. The sun gear S1 of the reduction planetary gear G1 is fixed to the transmission case 10 so that a support as a particular supporting member for fixing is not needed. Further, the hydraulic servo 30 of the first clutch C-1 is arranged on outer surface of the cylindrical boss portion 10b arranged in the front space of the reduction planetary gear G1, and the hydraulic servo 40 of the third clutch C-3 is arranged on outer surface of the front cylindrical portion 10f' arranged in the rear space of the reduction planetary gear G1. Therefore, the hydraulic paths for applying hydraulic pressures to the hydraulic servos of the both clutches are directly connected to the cylindrical boss portion 10b and the front cylindrical portion 10f' without through another shafts. As a result, the hydraulic paths which have to be filled with oil is shortened so that the responsiveness to the applications of the hydraulic pressures are improved. Further, the hydraulic pressure is directly applied from the cylindrical portion 10f' of the support wall 10A to the hydraulic servo 40 of the third clutch C-3 without through another rotational members. As a result, a number of seal rings for sealing the hydraulic path is reduced so that a sliding load occurred by the seal rings is reduced. Further, the first engaging member is a band brake needed an arrangement space which is pretty small in radial direction. Therefore, the radial outer space of the third clutch C-3 is obtained even by radially superposing the brake B-1 and the third clutch C-3. Then, the diameter of the clutch is increased in order to obtain the torque transmission capacity. As a result, a number of the structural elements of the frictional member 45 is reduced so that the axial size of the clutch is reduced.

[0073] In the first embodiment, the one-way clutch F-1 and the brake B-2 are arranged in the front side of the first clutch C-1, that is, in the mostly front position of the shift mechanism. However, these arrangement positions are properly changed. Fig. 5 illustrates the automatic transmission of the second embodiment, in which the one-way clutch F-1 and the brake B-2 are arranged in the front side of the support wall 10A, that is, in the rear side of the third clutch C-3, with a sectional view. Only differences with the first embodiment in this embodiment will be described for preventing a duplication of the description.

[0074] In this second embodiment, as aforementioned, the arrangements of the one-way clutch F-1 and the brake B-2 are changed comparing with the first embodiment. The hydraulic servo 80 of the brake B-2 is connoted with the support wall 10A. In this embodiment, it is not needed that the inner race of the one-way clutch F-1 is extended in the radial outer space of the first clutch C-1 for connecting to the drum 41 of the third clutch C-3. The inner race of the one-way clutch F-1 is connected to the inner portion of the drum 41 of the third clutch C-3 with the shortest line. Therefore, no member is arranged in the radial outer side of the drum 31 of the first clutch C-1 transmitting the output rotation of the reduction planetary gear G1. As a result, it is possible to detect the input rotation by detecting the rotation of the drum 31 at the radial outer side of the drum 31. In this embodiment, an input rotation sensor Sn is arranged on the surrounding wall portion of the transmission case 10.

[0075] In this embodiment, the clutch hub 46 of the third clutch C-3 is connected through the clutch drum 31 of the first clutch C-1 to the carrier C1 of the reduction planetary gear G1 so that no member, which is arranged for transmitting the output rotation of the reduction planetary gear G1 to the first and third clutches, is arranged in the radial outer space of the both clutches. Therefore, the diameters of the first and third clutches, which transmit the torque amplified at the reduction planetary gear G1, are able to be increased. As a result, it is easy to maintain the torque transmission capacity of the clutches according to the transmission torque without increasing the axial sizes of the clutches. Therefore, the transmission is compact. Further, the clutch drum 31 of the first clutch, which always rotates, is arranged in the mostly radial outer position. Therefore, it is easy to detect the input rotation needed for the transmission control without arranging the sensor Sn in the deep portion of the transmission case 10. Further, the front end wall portion 10a is not needed to be thick for arranging the sensor Sn so that the axial size of the transmission is reduced. Further, the axial size of the transmission is increased by arranging the support wall 10A, however, no member is extended in the radial outer space of the first and third clutches C-1, C-3 so that the diameter of the clutches are able to be increased for obtaining the torque transmission capacity, then, the axial sizes of the clutches are reduced. That is, the increase of the axial size of the transmission is counterbalanced with the reduction of the axial size of the clutches. As a result, the increase of the axial size of the transmission is minimized.

[0076] Further, in this embodiment, the hydraulic servo 80 of the second brake B-2 is connoted with the support wall 10A so that a particular member for the hydraulic servo cylinder of the second brake B-2 is not needed. Therefore, an increase of a number of the structural members is prevented.

[0077] Next, Fig. 6 illustrates the third embodiment, in which the one-way clutch F-1 and the brake B-2 are arranged in the rear side of the support wall 10A, that is, in the front side of the second one-way clutch F-2, with a sectional view. In this third embodiment, the change points are the connecting portion of the one-way clutch F-1 and the opening direction of the hydraulic servo of the brake B-2. In this case, the inner race of the one-way clutch F-1 is connected to the power transmission member 11E which connects the drum 31 of the third clutch C-3 to the sun gear S2 of the planetary gear set G. The hydraulic servo 80 of the brake B-2 is connoted with the support wall 10A and faced to the frictional member 85 arranged in the rear side of the hydraulic servo 80. Therefore, the one-way clutch F-1, the brake B-2 and the hydraulic servo are not arranged between the hydraulic servo 40 of the third clutch C-3 and the radial wall portion 10e so that the hydraulic path formed in the front cylindrical portion 10f' is shortened.

[0078] Fig. 7 illustrates the fourth embodiment which is different from the aforementioned embodiments and does not have the one-way clutch F-1 and the brake B-2. In this embodiment, regarding to the engaging control of the brake B-1, the hydraulic servo control of the band brake as the brake B-1 is complicated, however, it is pretty effective for reducing the axial size of the transmission.

[0079] In considering with the brake B-1, in case the one-way clutch F-1 and the brake B-2 are not arranged, it is effective that the brake B-1 is structured from a multi-disc type brake which is easier to control than a band brake. Fig. 8 illustrates the fifth embodiment in which the brake B-1 is changed to the multi-disc type. Another members are almost same with the fourth embodiment. In this embodiment, the frictional members 35, 45 of the first and third clutches C-1, C-3 and the frictional member 65 of the brake B-1 are axially aligned. Therefore, the frictional members 35, 45 of the first and third clutches C-1, C-3 are relatively moved to the front side so that the frictional member 35 of the first clutch C-1 is arranged in the radial outer position of the hydraulic servo 30 and the frictional member 45 of the third clutch C-3 is arranged in the radial outer position of the reduction planetary gear G1. The hydraulic servo 60 of the brake B-1 is connoted with the radial wall portion 10e of the support wall 10A and faced to the frictional member 65, that is, to front side. In this case, the separator plates of the frictional member 65 are engaged with the spline formed on the inner surface of the surrounding wall of the transmission case 10, and the frictional elements are supported on the drum 41 of the third clutch C-3 as the hub.

[0080] Fig. 9 illustrates the sixth embodiment in which the cylindrical portion of the support wall 10A is not arranged. Another members are almost same with the fourth embodiment. In this embodiment, a hydraulic path L_R for applying the hydraulic pressure to the hydraulic servo 40 of the third clutch C-3 is connected through a space as a hydraulic path, which is surrounded by the power transmission members 11D, 11E and bushes 11F, 11G, to the hydraulic servo 40. Then, a pair of seal rings, which are aligned in front and rear side, is arranged at the relative rotating portion between the support wall 10A and the power transmission member 11E for sealing. In this embodiment, the diameter of the hydraulic path connecting portion is reduced comparing with the case the cylindrical portion is arranged so that the affect of the centrifugal force is reduced when the hydraulic pressure is applied to the hydraulic servo 40. As a result, the control performance of the third clutch is improved.

[0081] In the aforementioned embodiments, the second clutch C-2 is arranged in the rear side of the planetary gear set G, however, the second clutch C-2 is able to be arranged in the next rear side position of the reduction planetary gear G1. Fig. 10 illustrates the seventh embodiment having thus arrangement with a schematic diagram, and Fig. 11 illustrates a specific cross sectional structure. As illustrated in Fig. 10, in this embodiment, the arrangement of the each element is same with the first embodiment excepting the arrangement of the second clutch C-2. Therefore, the same notations or reference numerals are put and the descriptions are omitted.

[0082] In this embodiment, as illustrated in Fig. 11, the input shaft 11 is different. The majority of the input shaft 11 is an intermediate shaft transmitting the input rotation transmitted from the clutch C-2. A rear portion 11B as the intermediate shaft is relatively rotatable with a front portion 11A, and the front end of the rear portion 11B is supported through a bearing by the front portion 11A. The hydraulic servo 50 of the second clutch C-2, which is arranged in the next rear position of the reduction planetary gear G1, is connoted with the rear end portion of the front portion 11A of the input shaft and the clutch drum 51 fixed on the rear end portion. The clutch hub 56 is connected to the front end of the rear portion 11B of the input shaft.

[0083] In this case, the cylindrical portion 10f of the support wall 10A is axially extended in back and forth. The hydraulic servo 40 of the third clutch C-3 and the inner race of the one-way clutch F-1 are rotatably supported on the front cylindrical portion 10f', and the inner race of the one-way clutch F-2 is fixed on the rear cylindrical portion 10f''. The hydraulic servo 80 of the brake B-2 and the hydraulic servo 70 of the brake B-3 are connoted with the radial wall portion 10e of the support wall 10A back to back with each other. The frictional member 55, which has small diameter because the torque transmission capacity of the second clutch C-2 transmitting

the non-decelerated rotation is relatively small, and the frictional member 45 of the third clutch C-3, of which the torque capacity is maintained by increasing the diameter, are radially superposed so that the axial size of the transmission is appreciably reduced.

[0084] In this embodiment, the clutch drum 51 of the second clutch C-2 is able to be common with a power transmission member connected to the reduction planetary gear G1. Therefore, a number of power transmission members which radially transverse in the shift mechanism so that the axial size of the shift mechanism is reduced.

[0085] Fig. 12 illustrates the eighth embodiment in which the hydraulic servo 70 of the brake B-3 is arranged in the hindmost of the shift mechanism as same as the first through sixth embodiments. Another members are same with the seventh embodiment.

[0086] Fig. 13 illustrates the arrangement of the ninth embodiment which does not have the one-way clutch F-1 and the brake B-2. Another members are almost same with the eighth embodiment. As compared with the eighth embodiment illustrated in Fig. 12 with the same contraction ratio, the axial size of the transmission is extremely reduced.

[0087] Fig. 14 illustrates the tenth embodiment in which the arrangement order of the first and third clutches, C-1, C-3 and the reduction planetary gear G1 is same with the first embodiment and the one-way clutch F-1 and the brake B-2 are arranged between the reduction planetary gear G1 and the third clutch C-3. In this case, the cylinder of the hydraulic servo 40 of the third clutch C-3 is connoted with the clutch drum 41 structuring the output side member of the third clutch. The cylinder is faced to opposite side to the support wall 10A. The clutch drum 41 is connected to the sun gear S2 of the planetary gear set G. In detail, the support wall 10A comprises the cylindrical portion 10f" axially extended to the rear side, and the hydraulic servo 40 connoted with the clutch drum 41 of the third clutch C-3 is supported on the outer surface of the cylindrical portion 10f". Effects and losses of this arrangement are same with the effects and losses of the second embodiment against the first embodiment. Further, the clutch drum 41 of the third clutch C-3 as the brake drum of the first brake B-1 is supported by the hydraulic servo 40 and the sun gear S2 at the both sides so that the brake drum is certainly prevented to waggle during the brake engagement.

[0088] Fig. 15 illustrates the eleventh embodiment in which the positional relationship between the first clutch C-1 and the reduction planetary gear G1 is opposite from the aforementioned tenth embodiment and, according to that, the direction of the first clutch C-1 is reversed. In this case, the reduction planetary gear G1 is arranged on the boss portion 10b extended from the front end wall portion of the transmission case 10, and the sun gear S1 is fixed on the boss portion 10b. The clutch drum 31 connoting the hydraulic servo 30 of the first clutch C-1 is supported on the front cylindrical portion 10f' axially extended from the support wall 10A to the front side. The effects of this arrangement, which are substantially different from the aforementioned each embodiment, is that the hydraulic path for applying the hydraulic pressure to the hydraulic servo 30 is not needed to be formed in the front end wall portion 10a, which is structured from the oil pump body and in which hydraulic paths are complicatedly formed, of the transmission case because the front end wall portion 10a is exclusively used for supporting the reduction planetary gear G1. Further, the hydraulic pressures are applied to the hydraulic servos 30, 40 of the first and third clutches from the support wall which is arranged in the next position of the both clutches. Therefore, the hydraulic paths to the hydraulic servos of the both clutches are shortened, and the lengths are approximately same. As a result, the responsiveness of the each clutch is improved and the control characteristics of the both clutches are accommodated.

[0089] Fig. 16 illustrates the twelfth embodiment in which the second clutch C-2 is arranged in the next rear side of the reduction planetary gear G1. Another members are same with the tenth embodiment. In this case, as same as the seventh embodiment, the clutch drum 51 of the second clutch C-2 is able to be common with the connecting member connected to the reduction planetary gear G1.

[0090] Fig. 17 illustrates the thirteenth embodiment in which the position of the first clutch is changed from the twelfth embodiment. In this case, the first clutch C-1 is moved to the support wall 10A side, and the hydraulic servo 30 is supported on the cylindrical portion 10f'. The effects of this arrangement is same with the effects of the eleventh embodiment.

[0091] The aforementioned each embodiment embodies the invention applied to a parallel type transmission for a front engine rear drive (FR) vehicle. Finally, two embodiments will be described by embodying the invention with a transverse type transaxle for a front engine front drive (FF) or rear engine rear drive (RR) vehicle.

[0092] Fig. 18 and Fig. 19 are illustrate the fourteenth embodiment. In this embodiment, as shown by comparing the schematic diagram of the gear train of the first embodiment illustrated in Fig. 1 with the schematic diagram of the gear train of this embodiment, the both embodiments have substantially same structure, however, according to being the transverse type, this embodiment has three axis structure in which the shift mechanism 1A is arranged on a first axis, a counter gear mechanism 1B is arranged on a second axis and a differential unit is arranged on a third axis. Further, a counter drive gear 19A as the output member of the ring gear R2 is arranged on the first axis instead of the output shaft.

[0093] The counter gear mechanism 1B comprises a counter driven gear 13 and a differential drive pinion

gear 14 which are fixed on a counter shaft 12. The counter driven gear 13 is engaged with the counter drive gear 19A at the axial middle portion. The differential drive pinion gear 14 is engaged with a differential ring gear 15 of the differential unit 1c at the mostly front portion of the shift mechanism. The differential unit 1C is fixed to the differential ring gear 15 and comprises a differential case 16 in which a differential gear is arranged. Both output shafts 17 of the differential gear are connected to vehicle wheels.

[0094] As illustrating the specific structure with the cross sectional view in Fig. 19, in this embodiment, the counter drive gear 19A is arranged in the other side of the support wall 10A to the third clutch C-3. The support wall 10A comprises the cylindrical portion 10f" for supporting the counter drive gear 19A. the counter drive gear 19A is rotatably supported through a bearing on the cylindrical portion 10f".

[0095] In this embodiment, the axial size of the transmission mechanism is particularly limited comparing with the aforementioned each embodiment because of the vehicle mounting characteristic. Therefore, the combination of the one-way clutch and the brake, which is arranged in parallel with the brake B-1, is not used as same as the fourth embodiment. In this embodiment, the arrangement relationship and the connection relationship of the planetary gear set G, reduction planetary gear G1, the first through third clutches (C-1 through C-3), and the brakes (B-1 through B-3) and the one-way clutch F-2 as the first and second engaging members are substantially same with the fourth embodiment.

[0096] As detail structures, the second clutch C-2 is arranged on the boss portion 10d of the transmission case 10 so that the hydraulic pressure is applied to the hydraulic servo 50 without through the input shaft 11. According to that, the frictional member 55 of the second clutch C-2 is moved to the front side and arranged in the radial outer side of the planetary gear set G. The hydraulic servo 70 of the brake B-3 is arranged in the radial outer side of the hydraulic servo 50 of the second clutch C-2 and built in the rear end wall portion 10c of the transmission case 10.

[0097] Fig. 20 illustrates the fifteenth embodiment in which the order of all members of the shift mechanism is opposite from the fourteenth embodiment. In this embodiment, it is difficult that the hydraulic servo 70 is built in the front end wall portion 10a, which ordinary structured from the oil pump body, of the transmission case 10 because of the hydraulic paths arrangement. Therefore, the hydraulic servo 70 of the brake B-3 is attached to the surrounding wall of the transmission case 10 in the radial outer side of the planetary gear set G. Therefore, the frictional member 55 of the second clutch C-2, which is moved from the radial outer side of the planetary gear set G to the front side, is able to be radially enlarged and the hydraulic servo 50 is also able to be radially enlarged by using the radial outer space of the frictional member 55. As a result, a number of the structural members of the frictional member 55 is reduced so that the axial size of the clutch is reduced. That is, the axial size of the transmission is prevented to increase.

[0098] Thus, the invention is described based on the multiple embodiments. The invention should not be limited to the foregoing embodiments but can be modified in various manners based on its gist, and these modifica tion should not be excluded from the scope of the invention.

[0099] An automatic transmission comprises a reduction planetary gear , two clutches transmitting a decelerated rotation transmitted from the reduction planetary gear, and a planetary gear set comprising two elements to which the decelerated rotation is inputted through the two clutches. The reduction planetary gear and the two clutches are arranged in one side of the planetary gear set. the clutch is closer to the planetary gear set than the clutch. The input side member of the clutch is connected to an output element of the reduction planetary gear and the input side member of the clutch. The output side member of the clutch is extended through the radial inner space of the clutch and connected to one of the two elements of the planetary gear set. Then, no member is arranged in the radial outer side of the both clutches, and the both clutches are radially enlarged.

Claims

1. An automatic transmission establishing multistage gear ratios for a vehicle comprising:

   a reduction planetary gear;
   a first clutch and a third clutch which transmit a decelerated rotation transmitted through at least the reduction planetary gear; and
   a planetary gear set which is inputted the decelerated rotation transmitted through the first and third clutches;
   wherein the reduction planetary gear and the first and third clutches are arranged in one side of the planetary gear set;
   the first and third clutches input the decelerated rotation transmitted through the reduction planetary gear to two different elements of the planetary gear set;
   the third clutch is arranged in closer position to the planetary gear set than the first clutch;
   an input side member of the first clutch is connected to an output element of the reduction planetary gear and an input side member of the third clutch; and
   an output side member of the first clutch is extended through an inner space of the third clutch and connected to one of the two elements of the planetary gear set.

2. The automatic transmission for a vehicle according to claim 1 wherein one element of the reduction planetary gear is fixed to a boss portion extended from the transmission case.

3. The automatic transmission for a vehicle according to claim 2 wherein the transmission case comprises a support wall, the first clutch is arranged on the boss portion, and the third clutch is arranged in a neighbor position of the support wall.

4. The automatic transmission for a vehicle according to claim 3 wherein the third clutch is supported on a first cylindrical portion axially extended from the support wall.

5. The automatic transmission for a vehicle according to claim 4 wherein the input side member of the first clutch is a clutch drum, of which a hydraulic servo is formed in the inside, and arranged in order that a cylinder of the hydraulic servo opens to the reduction planetary gear.

6. The automatic transmission for a vehicle according to claim 4 wherein the output side member of the third clutch is a clutch drum, of which a hydraulic servo is formed in the inside, arranged in order that a cylinder of the hydraulic servo opens to the opposite side against the support wall, and connected to one element of the planetary gear set.

7. The automatic transmission for a vehicle according to claim 6 wherein the first engaging member is structured from a band brake in which the outer surface of the clutch drum of the third clutch is an engaging surface engaged by a band.

8. The automatic transmission for a vehicle according to claim 3 wherein the output member is a counter gear, and the counter gear is arranged in the other side of the support wall against the third clutch.

9. The automatic transmission for a vehicle according to claim 8 wherein the support wall comprises a second cylindrical portion supporting the counter gear.

10. The automatic transmission for a vehicle according to claim 2 wherein the transmission case comprises a support wall, the first clutch is arranged in one side of the support wall, and the third clutch is arranged in the other side.

11. The automatic transmission for a vehicle according to claim 10 wherein the first and third clutches are supported on a cylindrical portion axially extended from the support wall.

12. The automatic transmission for a vehicle according to claim 11 wherein the input side member of the first clutch is a clutch drum, of which a hydraulic servo is formed in the inside, and arranged in order that a cylinder of the hydraulic servo opens to the reduction planetary gear.

13. The automatic transmission for a vehicle according to claim 11 wherein the output side member of the third clutch is a clutch drum, of which a hydraulic servo is formed in the inside, arranged in order that a cylinder of the hydraulic servo opens to the opposite side against the support wall, and connected to one element of the planetary gear set.

14. The automatic transmission for a vehicle according to claim 13 wherein the first engaging member is structured from a band brake in which the outer surface of the clutch drum of the third clutch is an engaging surface engaged by a band.

15. The automatic transmission for a vehicle according to claim 1 wherein a first element of the planetary gear set is connected to the output side member of the first clutch,

> a second element is connected to an output side member of the third clutch and is possible to engage with a transmission case with a first engaging member,
> a third element is connected an output side member of a second clutch, which is inputted a non-decelerated rotation, and is possible to engage with a transmission case with a second engaging member, and
> a fourth element outputs shift rotations.

16. The automatic transmission for a vehicle according to claim 15 wherein the second clutch is arranged in neighbor position of the reduction planetary gear.

17. The automatic transmission for a vehicle according to claim 16 wherein the frictional member of the second clutch is arranged in the inner side of the frictional members of the first and third clutches.

18. The automatic transmission for a vehicle according to claim 16 wherein an input side member of the second clutch is a clutch drum, and the clutch drum is an input member inputting a rotation to the reduction planetary gear.

19. The automatic transmission for a vehicle according to claim 16 wherein the output side member of the second clutch is a clutch drum, the hydraulic servo is arranged in order that the cylinder opens to the reduction planetary gear side, and the clutch drum is connected through an intermediate shaft to the

third element of the planetary gear set.

20. The automatic transmission for a vehicle according to claim 19 wherein a frictional member, which connects between the input side member and the output side member, of the second clutch is arranged in the radial outer position of the reduction planetary gear, and the hub is integrated with an input member of the reduction planetary gear.

FIG. 1

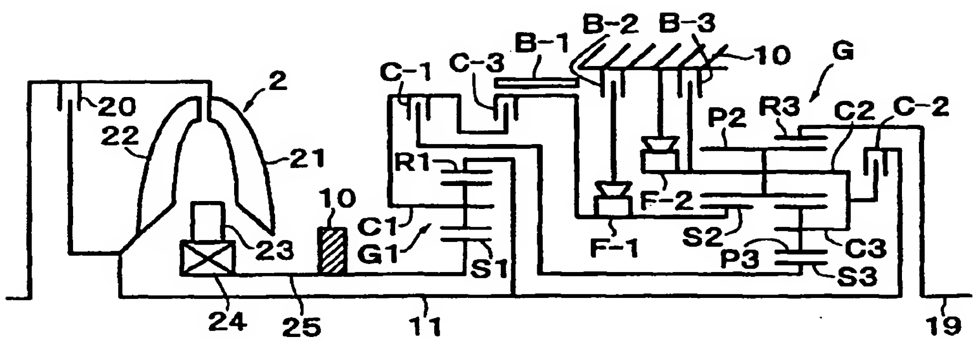

2
20
22
21
23
24
25
10
11
C-1
C-3
B-1
B-2
B-3
10
G
R1
C1
G1
S1
P2
R3
C2
C-2
F-2
F-1
S2
P3
C3
S3
19

FIG. 2

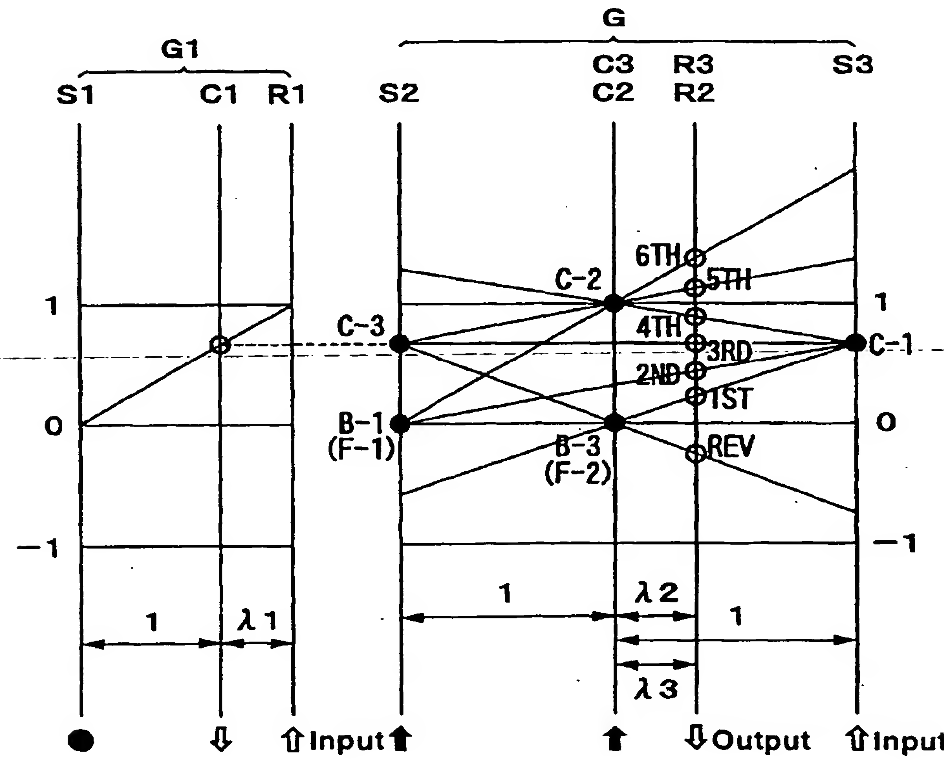
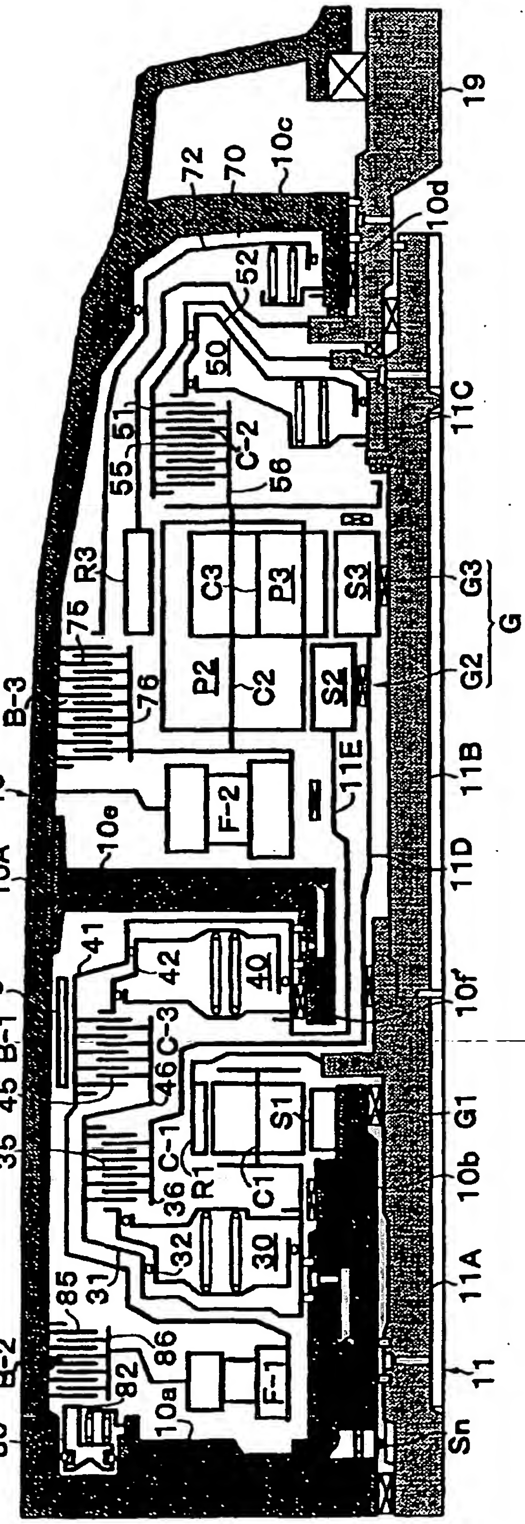

G1
S1
C1
R1
G
S2
C3
C2
R3
R2
S3
1
0
−1
6TH
5TH
4TH
3RD
2ND
1ST
REV
C-2
C-3
C-1
B-1
(F-1)
B-3
(F-2)
1
λ1
1
λ2
1
λ3
Input
Output
Input

FIG. 3

	C−1	C−2	C−3	B−1	B−2	B−3	F−1	F−2	Gear Ratio	Step
P										
REV			○			○			3.394	
N										
1ST	○					△		○	4.148	
2ND	○			△	○		○		2.370	1.75
3RD	○		○		●				1.556	1.52
4TH	○	○			●				1.155	1.35
5TH		○	○		●				0.859	1.34
6TH		○		○	●				0.686	1.25

FIG. 4



19
10c
70
72
10d
52
50
11C
51
55
C-2
56
R3
C3
P3
S3
G3
G
75
76
P2
C2
S2
G2
B-3
11E
11B
10
F-2
10e
11D
10A
41
42
40
6
10f
B-1
C-3
46
45
S1
G1
35
C-1
36
R1
C1
10b
32
30
31
11A
85
86
B-2
82
F-1
11
10a
80
Sn

FIG. 5

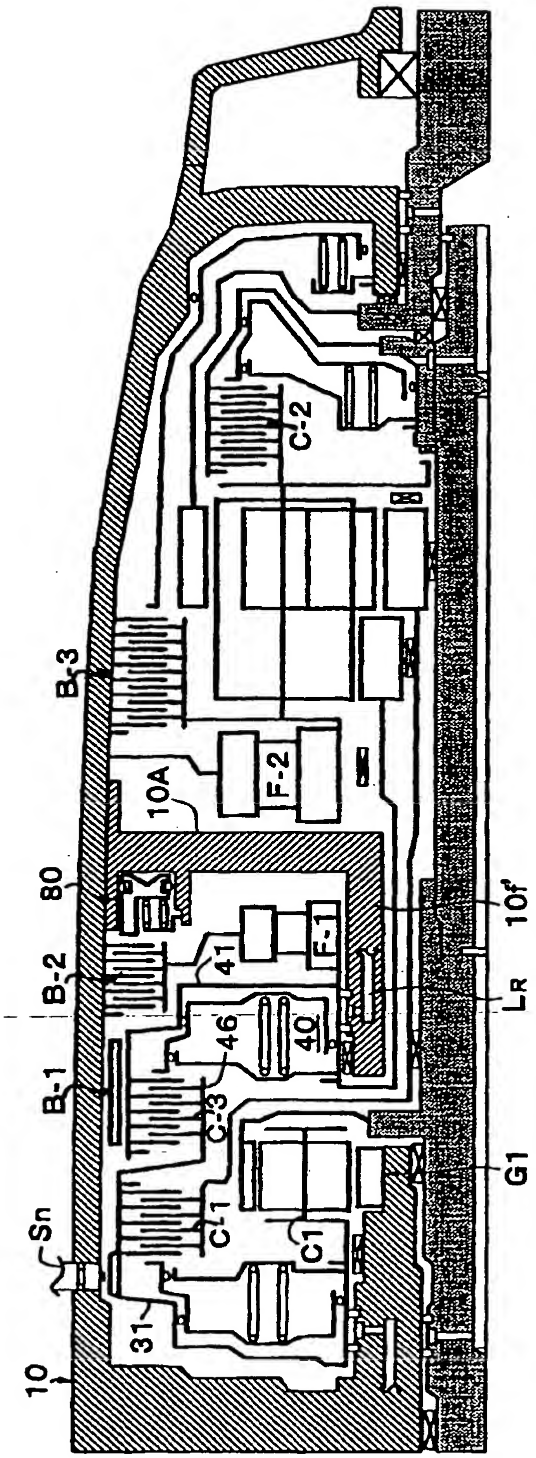


FIG. 6

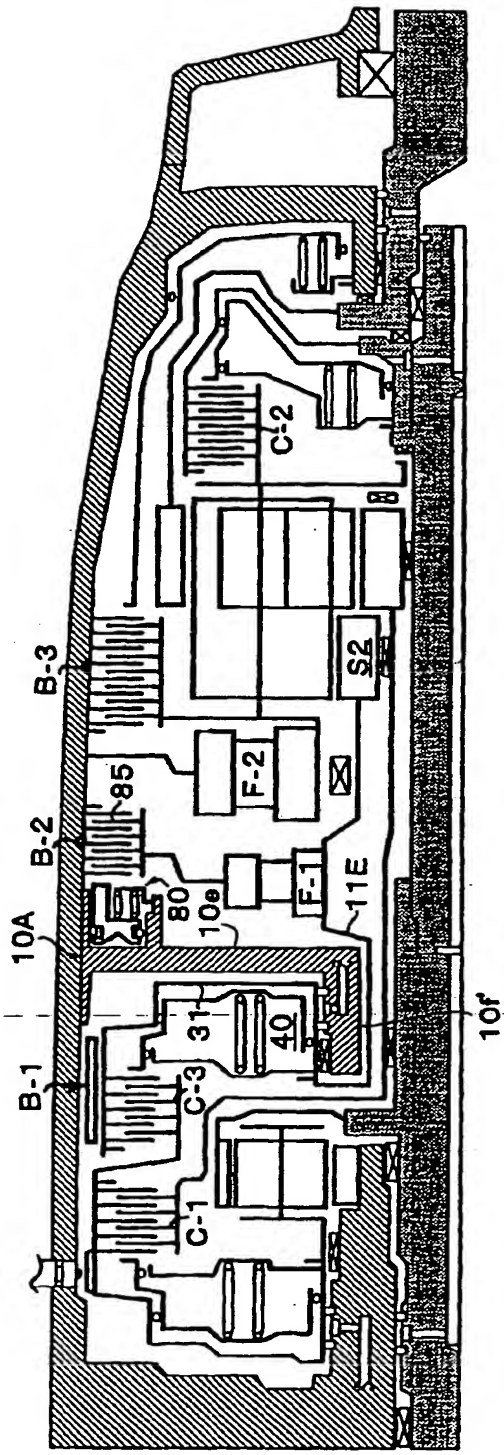


FIG. 7

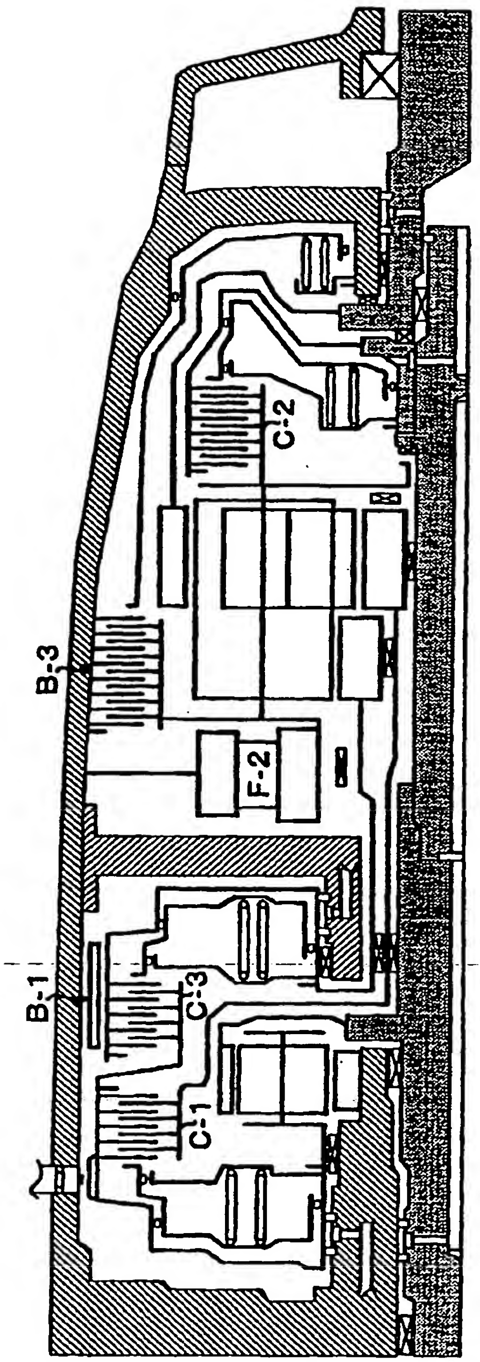
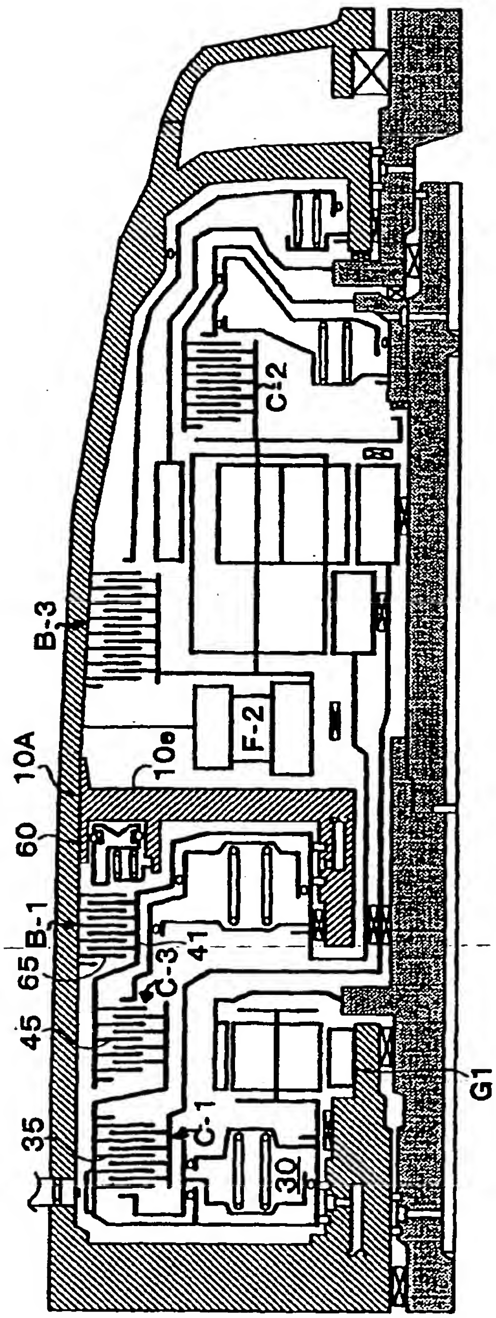


FIG. 8



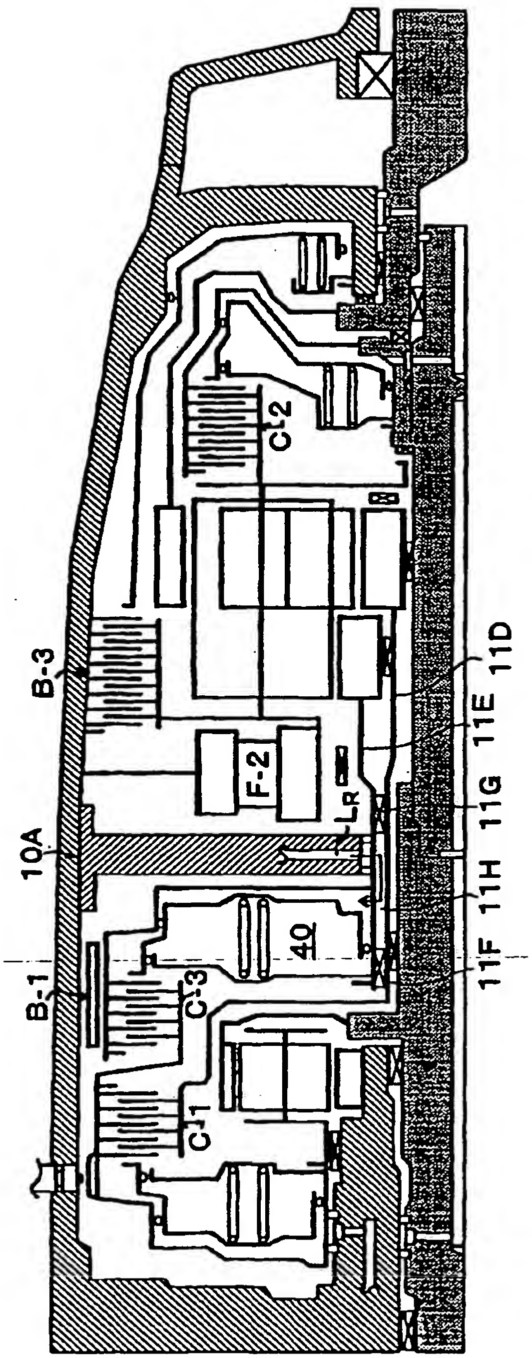


FIG. 9

FIG. 10

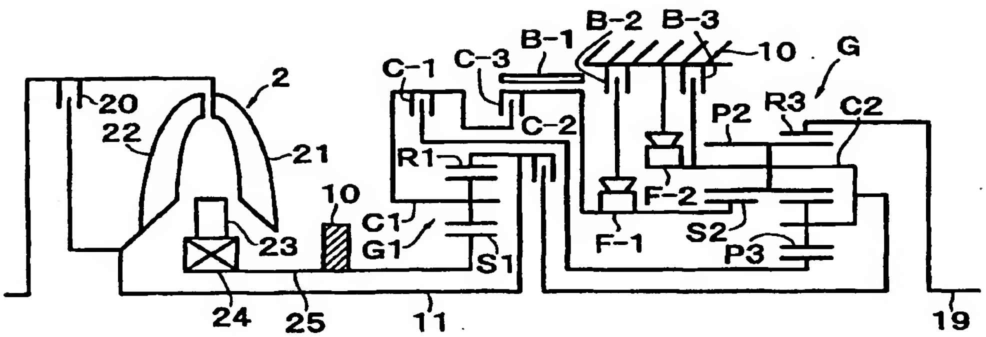

2
20
22
21
23
24
25
10
C-1
C-3
B-1
B-2
B-3
10
G
C-2
R1
C1
G1
S1
11
F-1
F-2
P2
R3
C2
S2
P3
19

FIG. 11

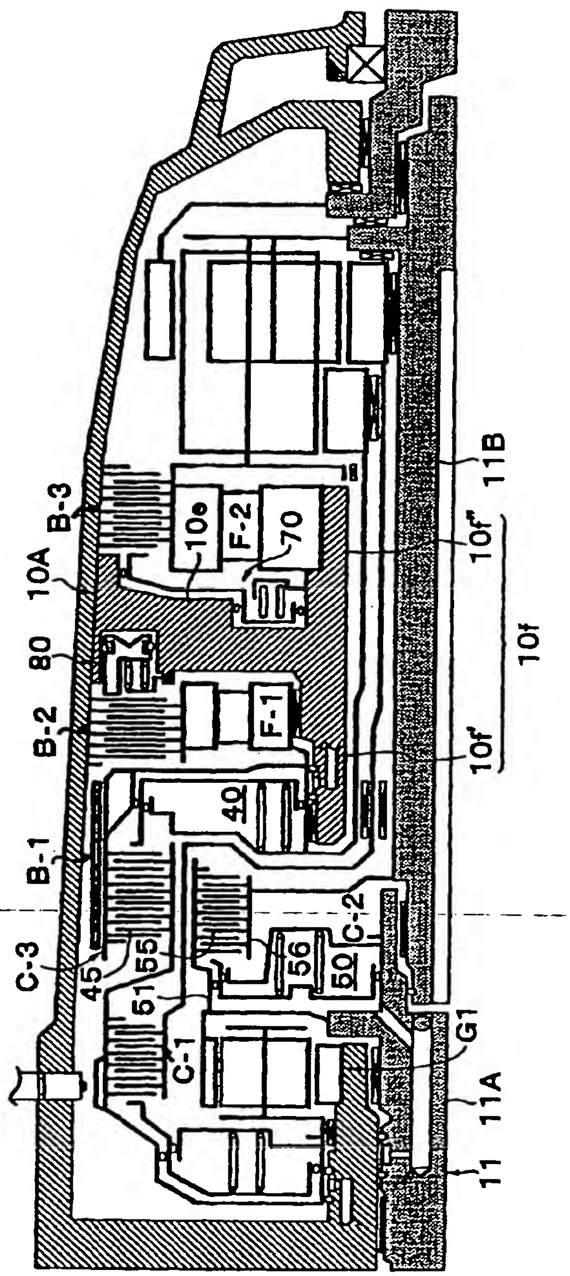

B-3
10e
F-2
70
10A
80
B-2
F-1
10f'
10f
10f''
11B
40
B-1
C-2
C-3
45
55
56
50
51
C-1
G1
11A
11

FIG. 12

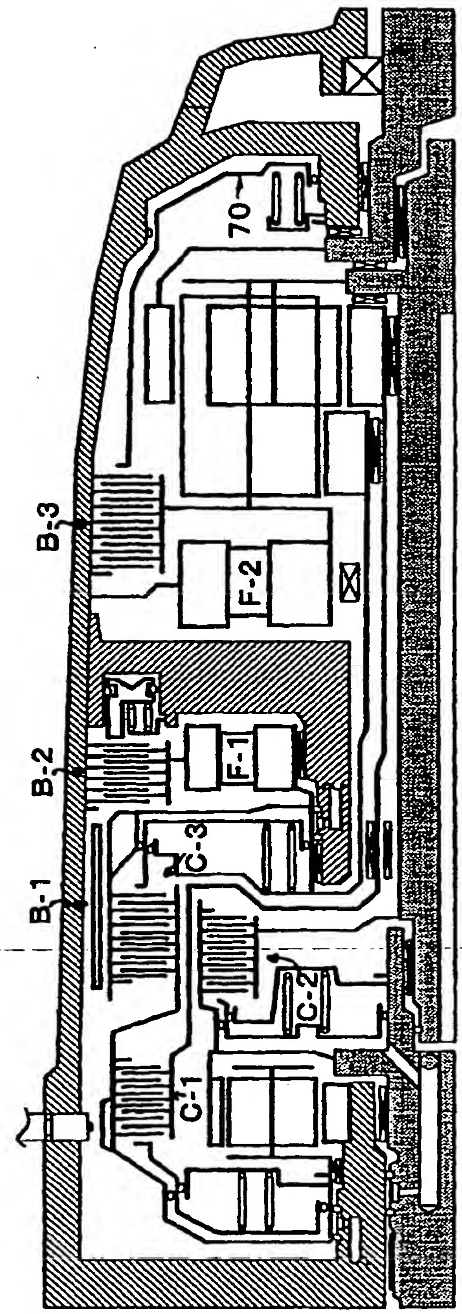


FIG. 13

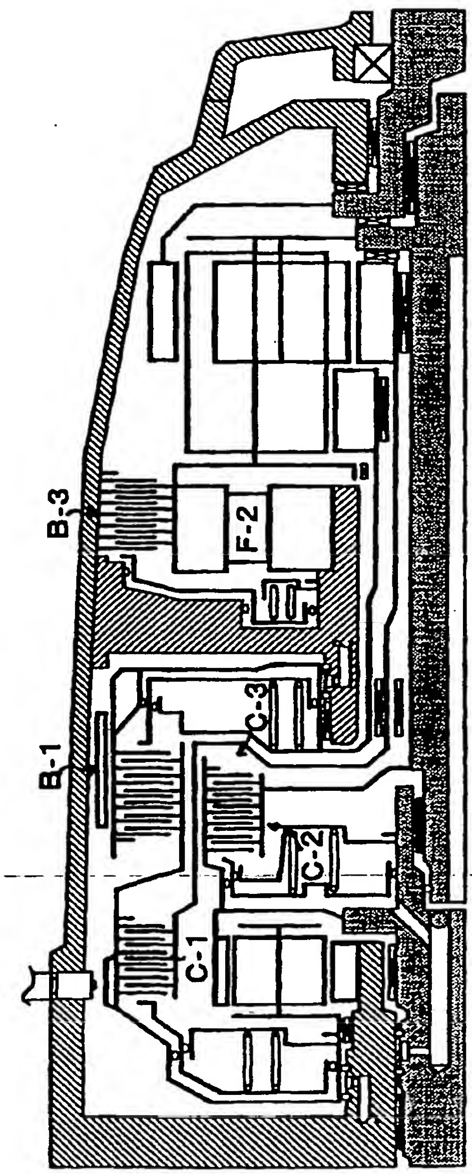


FIG. 14

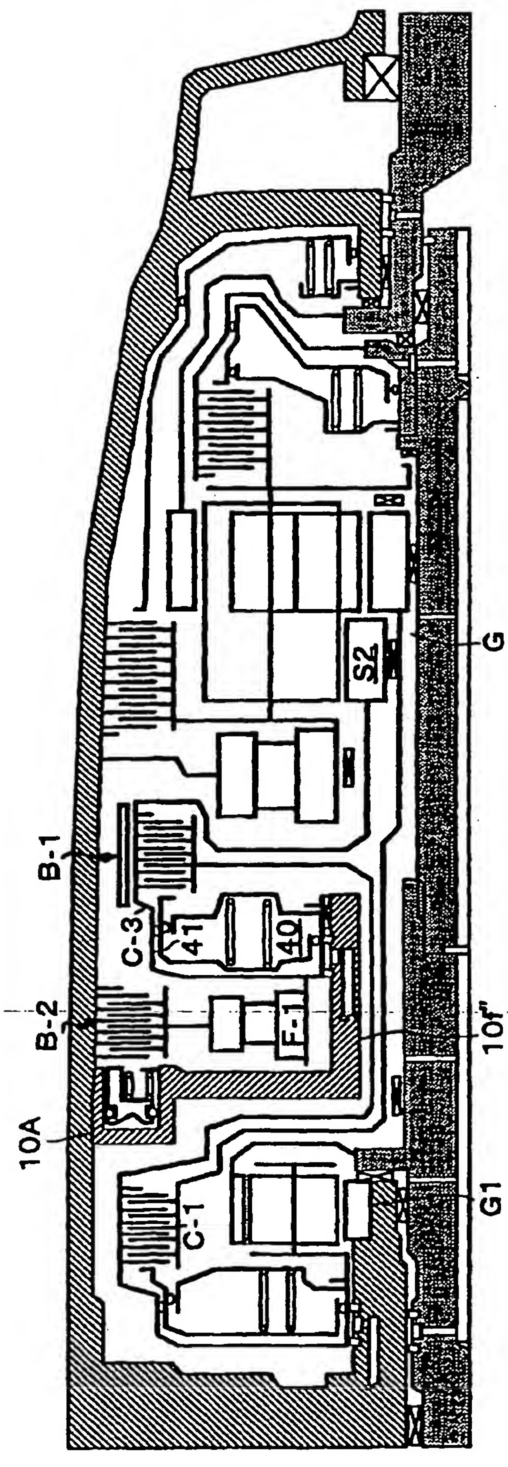


FIG. 15

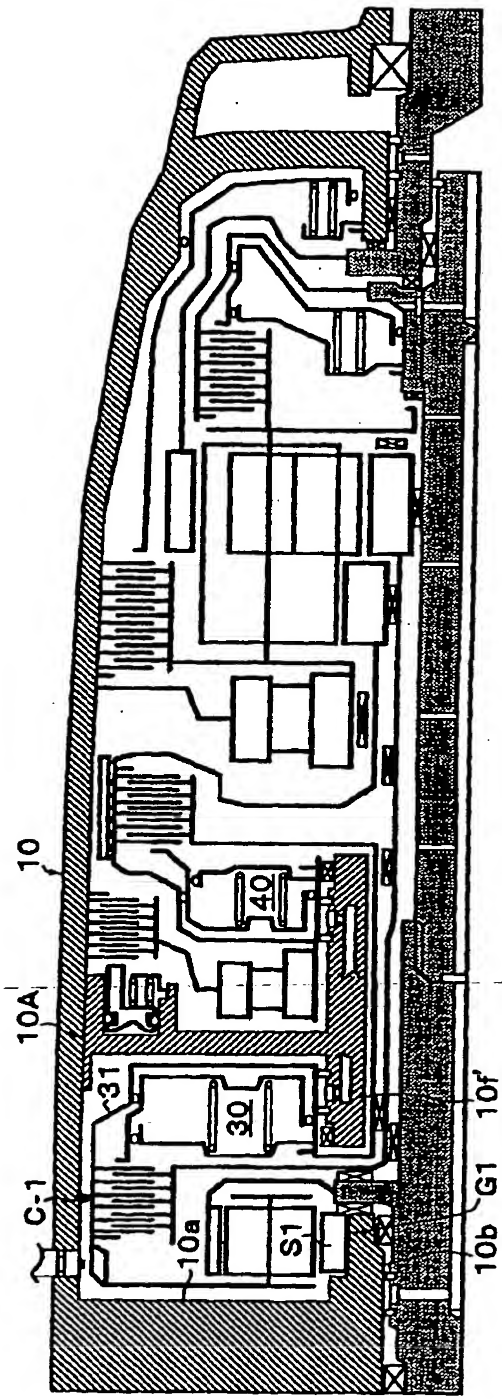


FIG. 16

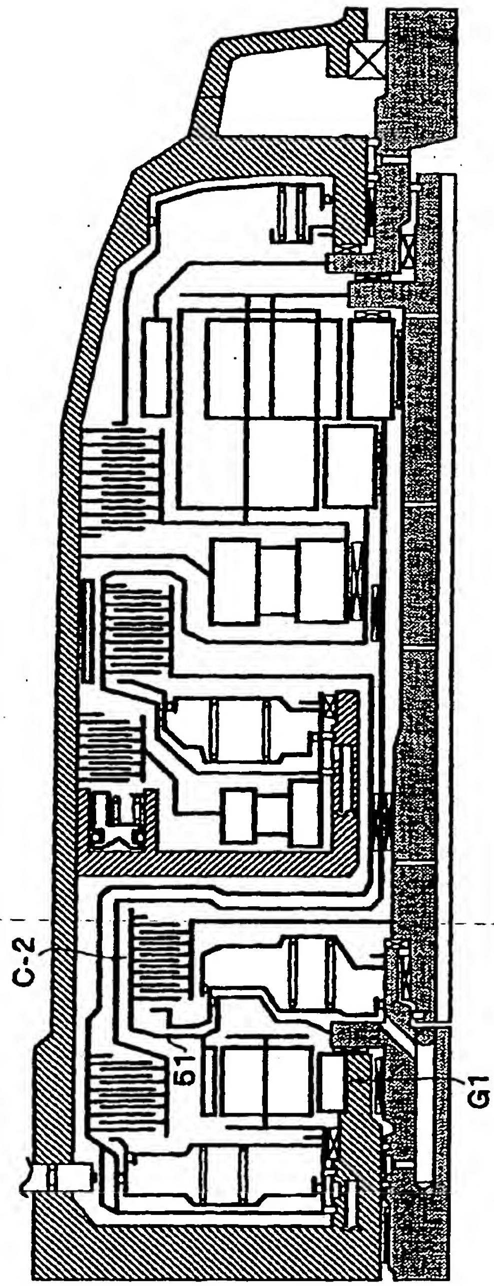


FIG. 17

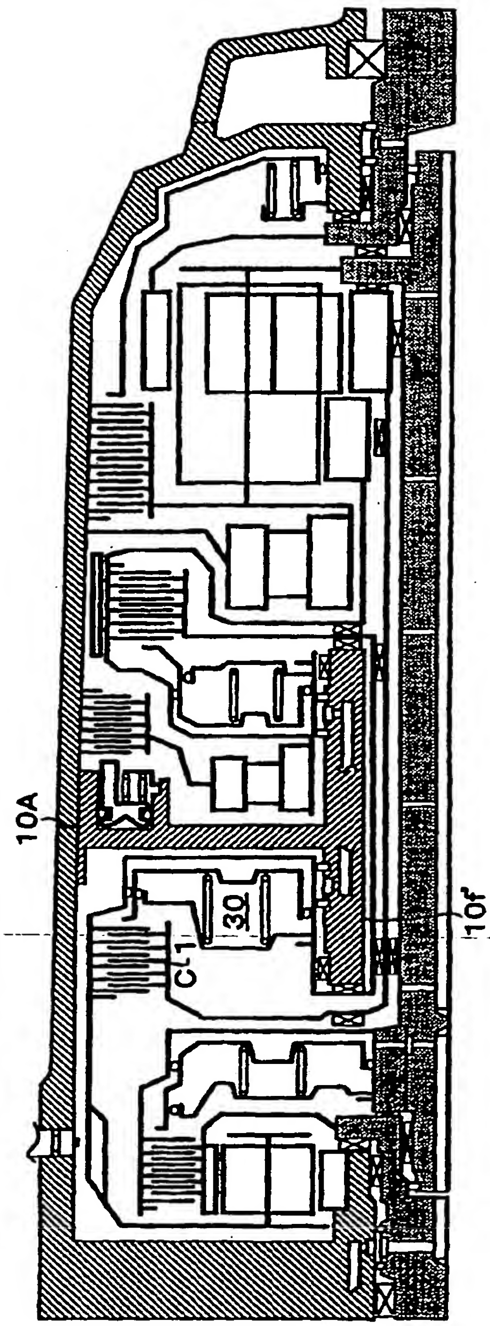


FIG. 18

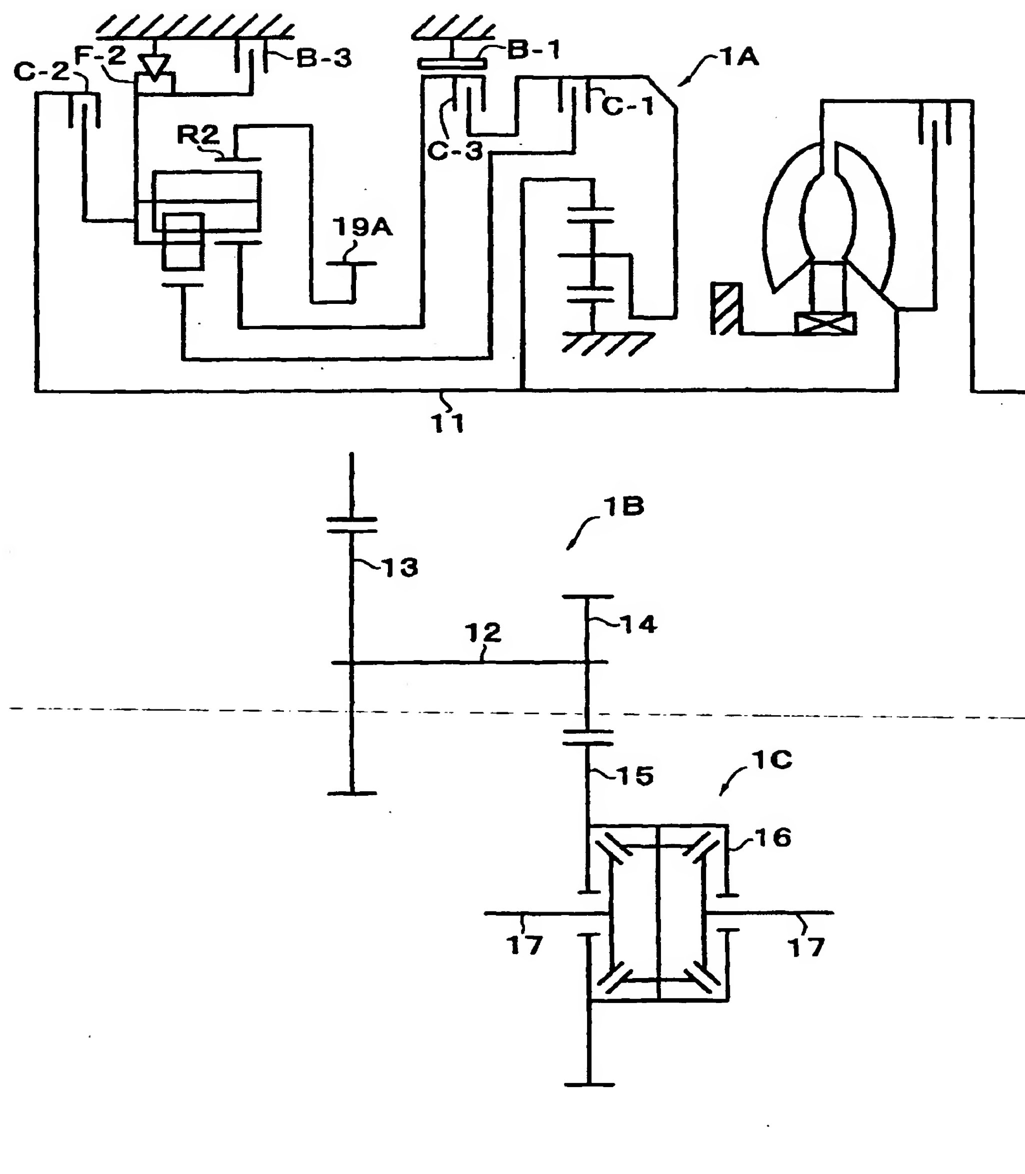


FIG. 19

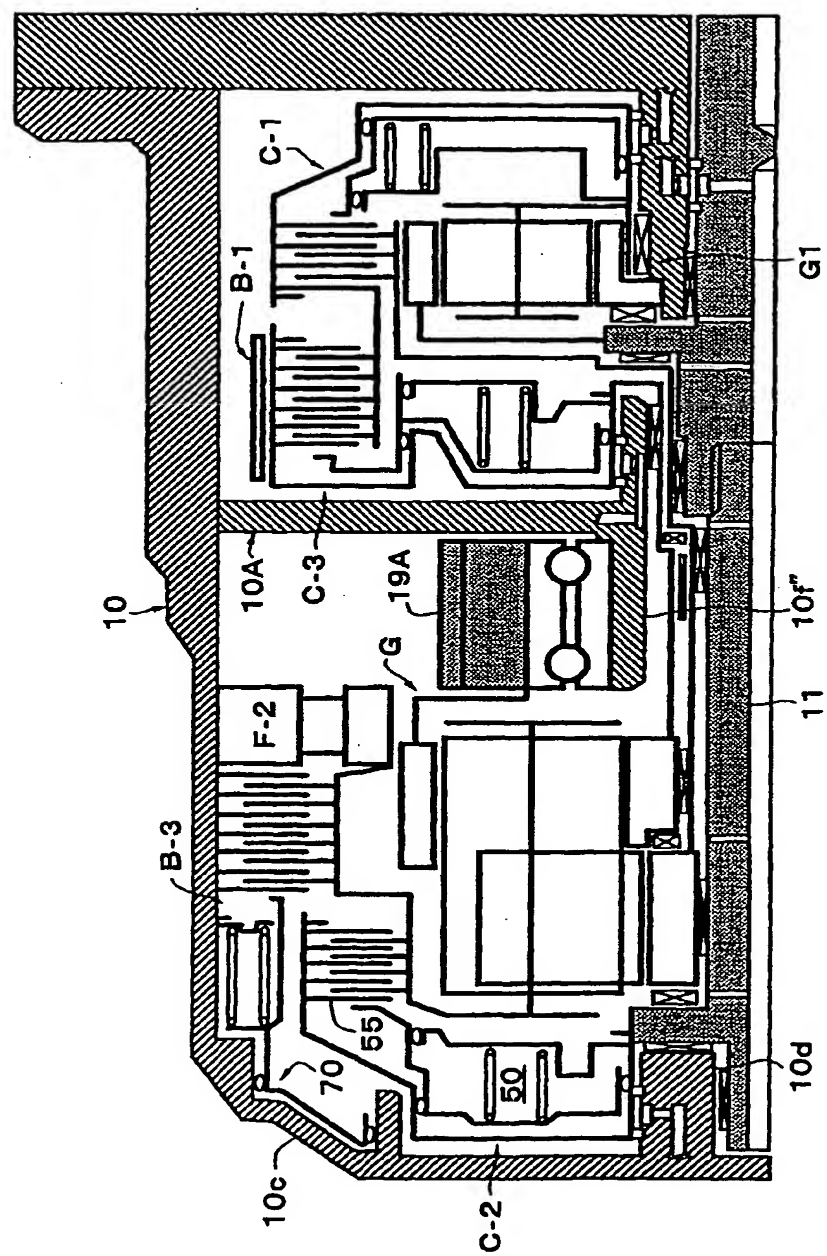


FIG. 20